The Vibration of a Drum Type Washing Machine at Different Speeds

we want to study :

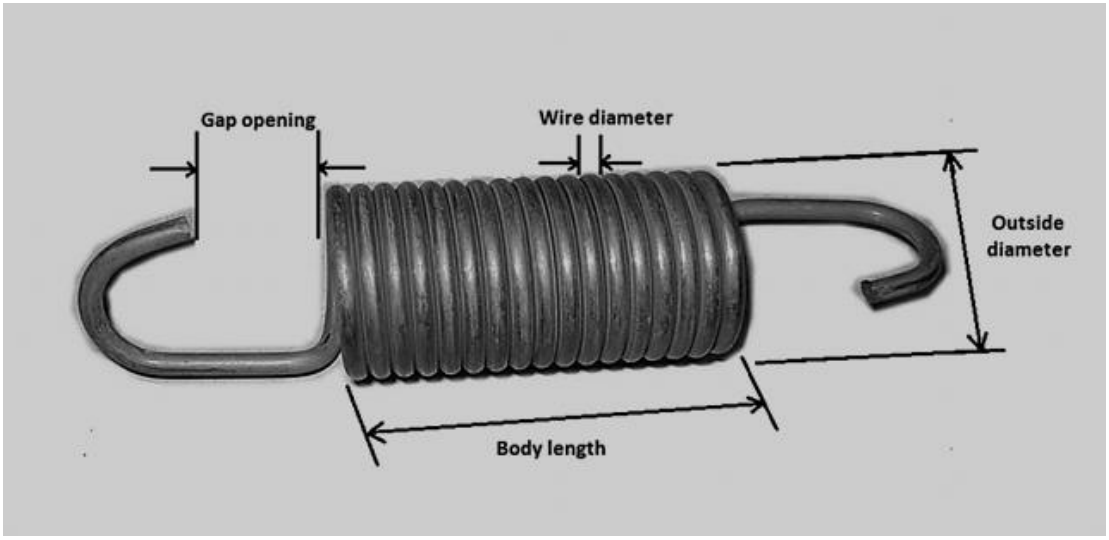
1* The effect of the Isolators stiffness, damping coefficient and the drum mass for specific laundry capacity

<u>Aims</u> : decrease vibration and noise excited by the rotating unbalance through the spinning cycle, as well increase the isolate performance

<u>1) Experimental part :</u>

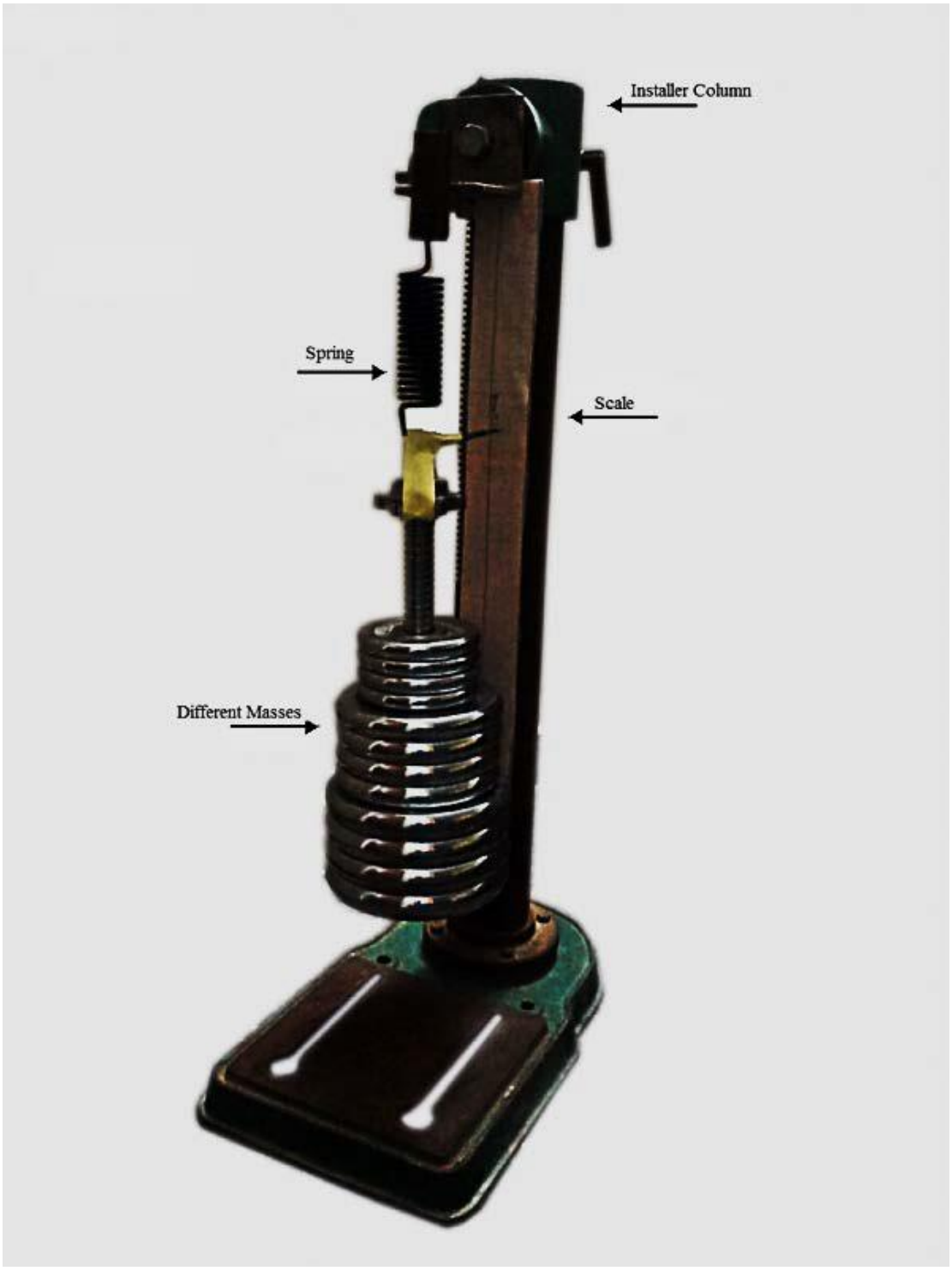
Washing machine used in examples is with wash capacity 8 Kg and dry capacity 5 Kg dimension 600 mm width, 560 mm depth and 850 mm height

1) Procedure of Measuring Spring Coefficient (K):



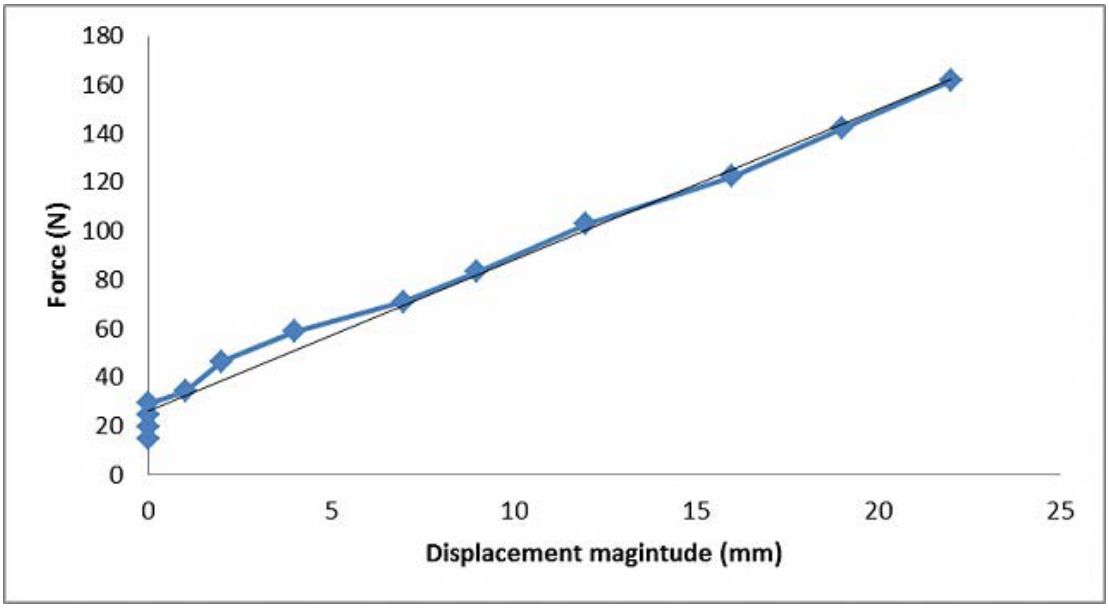
Electronics Washing Machine Hinge Spring

The spring was hanged on the installer column and after increasing masses on the spring, the extension of spring was measured after the addition of each mass, this device is shown in :



Examination device of the spring coefficient

This Figure shows a relation between the displacement magnitudes of hinge spring with the affected forces for each mass added



Force – displacement curve for Hinge Spring of washing machine

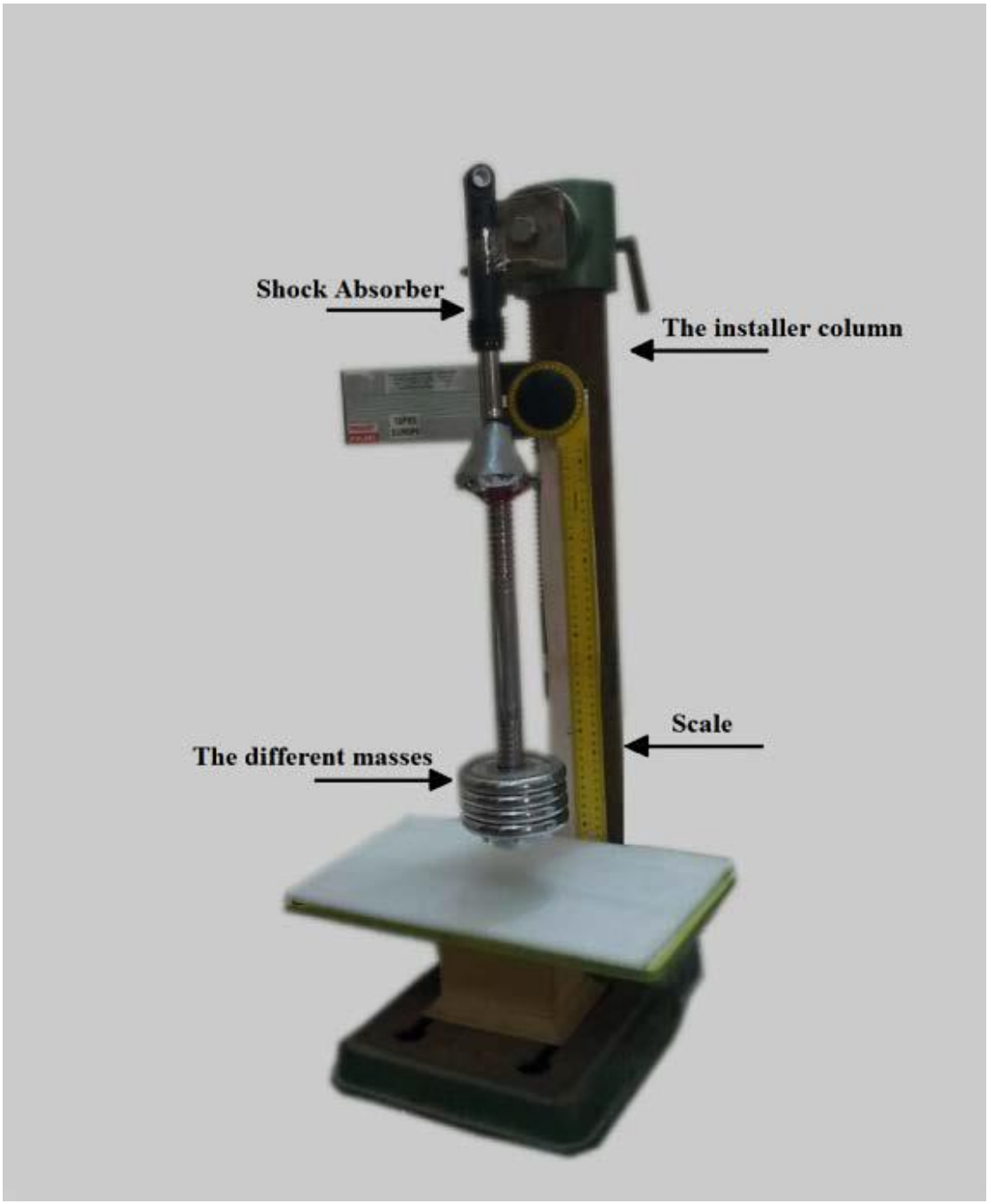
The constant stiffness (k) is calculated:

$K=\Delta y/\Delta x=5880$ N/m

2) Procedure of Measuring damping coefficient(c)

The aim of experiment is to measure the damping coefficient (c) for Shock Absorber of washing machine, as shown in Figure

Washing Machine Shock Absorber

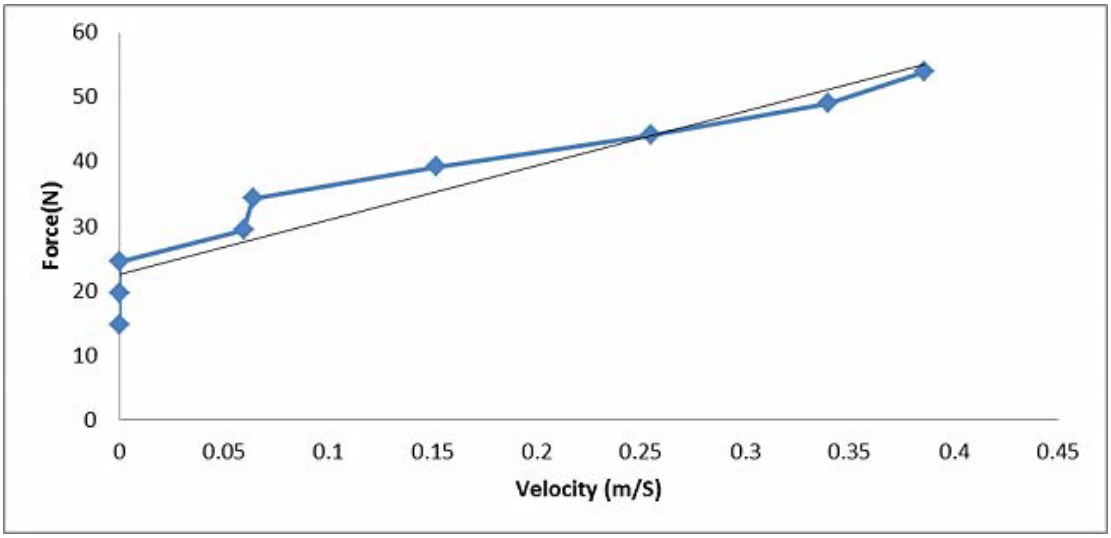


Examination device of the damping coefficient

Method of Measurement:

A. Suspended the damper on the installation column.
B. Determination a certain distance from the tip end of the damper until the surface level.
C. suspend the blocks on the free end of the damper.
D. Adjustment the stopwatch to calculate the time it takes to reach the mass of the surface level at every addition of the block.

This Figure shows a relation between the velocities of shock absorber with affected forces for of each mass addition



Force – velocity curve for shock absorber of washing machine

The constant coefficient (c) that is calculated:

$$c=\Delta F/\Delta V = 74.8 \text{ N.s/m}$$

To measure the Velocity and Displacement, A device it has been used (VIBRATION METERS) Where Vibration sensors are installed sites in the middle of each panel in the right and left and upper and above the washing machine base, as shown

Installation locations of Vibration sensors

Unbalanced masses were attached to the inside of the spin drum. Instead of attaching clothes to the inner wall of the spin drum in the spin-drying stage, Pieces of iron plates were used. Pieces of iron installed by magnets of the type Super Strong Round Disc Magnets, Figure down illustrates the picture of where the mass is installed in a drum of washing machine by Magnets

Unbalance mass site

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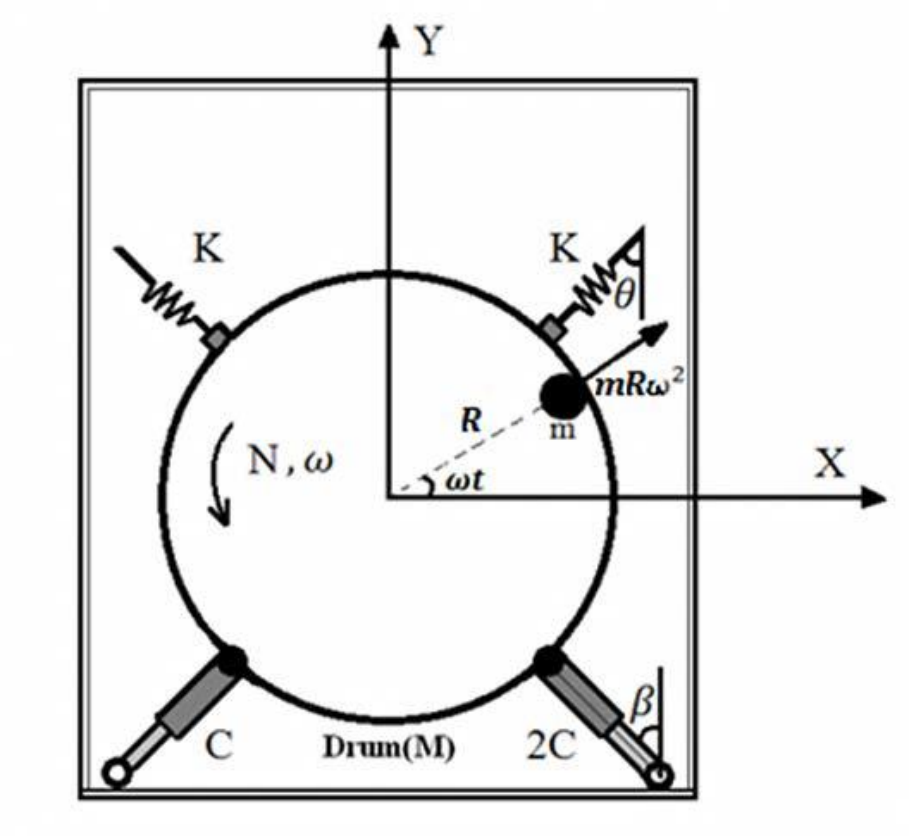
2) Theoretical Part

The system can be reduced in to simple physical diagram system as shown in figure

Balancing involves placing correction masses on to the drum, so that the centrifugal forces due to these masses cancel out those caused by the inherit unbalanced mass, thus canceling out vibration.



Physical Model of a Front load washing machine

Illustration of suspension parts of washing machine

**** Drum Vibration analysis during spinning**

By the basics of force stability and using the Newton second law of motion the equation of motion can be written as follows:

$$\Sigma Fx = (M - m)\,\ddot{x} \qquad (1)$$

M = Total mass of the drum

m = unbalanced mass

On more simplistic we get the equation given below:

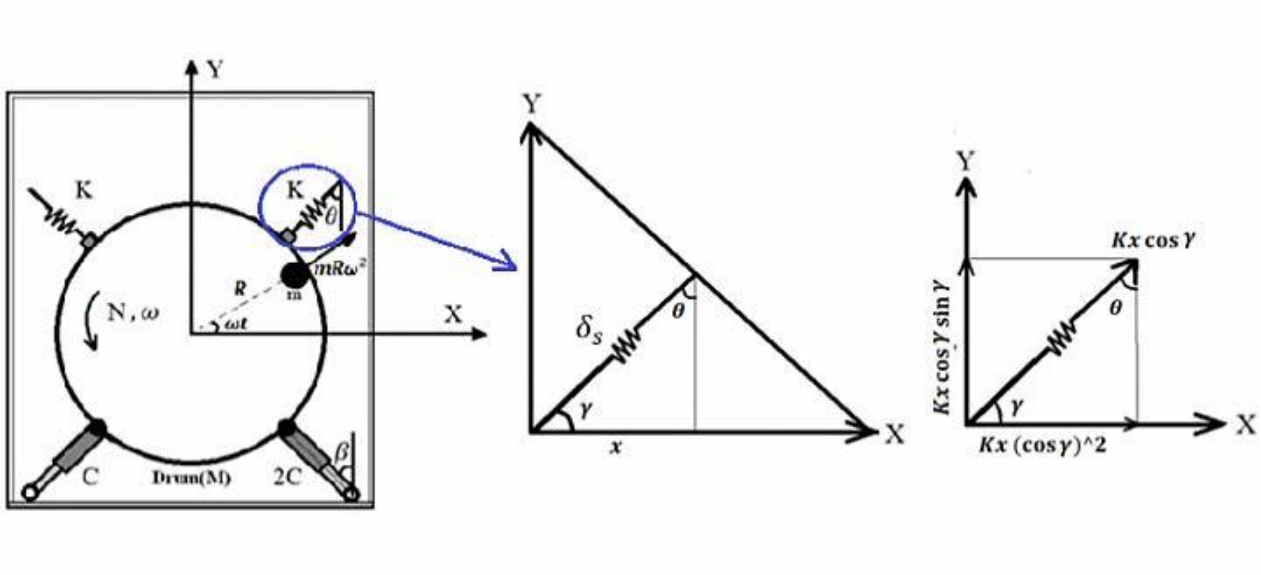
$$M\ddot{x} + c_e\dot{x} + k_e x = Fx \qquad (2)$$

‘Where

$$Fx = Fo\cos\omega t = mR\omega^2\cos\omega t \qquad (3)$$

This force will have an orientation ωt with the x-axis; the inertia force Fo will have an x-component

The dynamic model of the system can be derived in terms of the dynamic motion x and y in the two directions (x and y). However, it is possible to perform simpler analysis by considering only the motion in one direction (say x). The drum dynamic motion in the horizontal direction is x [1]. The deflection in the spring δs depends on its inclination angle θ as shown in Figure :



Representation deflection in the spring

$\delta s = x\cos\gamma = x\sin\theta$ (4)

In the same way, the deflection of the damper piston δd is:

$\delta d = x\cos\gamma = x\sin\beta$ (5)

The differential equation of the drum is obtained as:

$M\ddot{x} + 3c(\sin\beta)^2\dot{x} + 2k(\sin\theta)^2 x = m\mathrm{R}\omega^2\cos\omega t$ (6)

Where,

$\dot{x}$: is the vibration velocity & $\ddot{x}$: is the vibration acceleration

With: $c_e = 3c\,(\sin\beta)^2$, $k_e = 2k\,(\sin\theta)^2$

In particular, the response is harmonic and has the same frequency as the excitation frequency. Hence, a solution of Eq. (6) can be assume in the form

$$x = A \cos \omega t + B \sin \omega t \qquad (7)$$

In which A and B are constants yet to be determined, Inserting Eq. (7) into Eq. (6) gives,

$$M(-A\omega^2 \sin\omega t - B\omega^2 \cos\omega t) + c_e(-A \sin\omega t + B \cos\omega t) + k_e(A\cos\omega t + B \sin\omega)$$

$$= Fo \cos\omega t \qquad (8)$$

Equation (8) can be satisfied provided the coefficients of $\sin\omega t$ on both sides of the equation are equal, and the same can be said about the coefficients of $\cos\omega t$, which yields two algebraic equations in A and B as follows:

$$(k_e - M\omega^2)A + c_e\omega B = 0 \qquad (9)$$

$$(k_e - M\omega^2)B + c_e\omega A = Fo \qquad (10)$$

Using Cramer's rule, the solution of Eq. (9) and (10) is:

$$A = \frac{-c_e \omega F_o}{(k_e - M\omega^2)^2 + (c_e \omega)^2} \qquad (11)$$

$$B = \frac{(k_e - M\omega^2) F_o}{(k_e - M\omega^2)^2 + (c_e \omega)^2} \qquad (12)$$

Introducing Equations (11) and (12) in Eq. (7), the steady-state solution can be obtained as follows:

$$x = \frac{F_o}{\sqrt{(k_e - M\omega^2)^2 + (c_e \omega)^2}} \sin(\omega t - \gamma) \qquad (13)$$

Otherwise $\omega_n = \sqrt{\frac{2K}{M}}$, $\xi = \frac{3c}{2M\omega_n}$ $F_o = mR\omega^2$

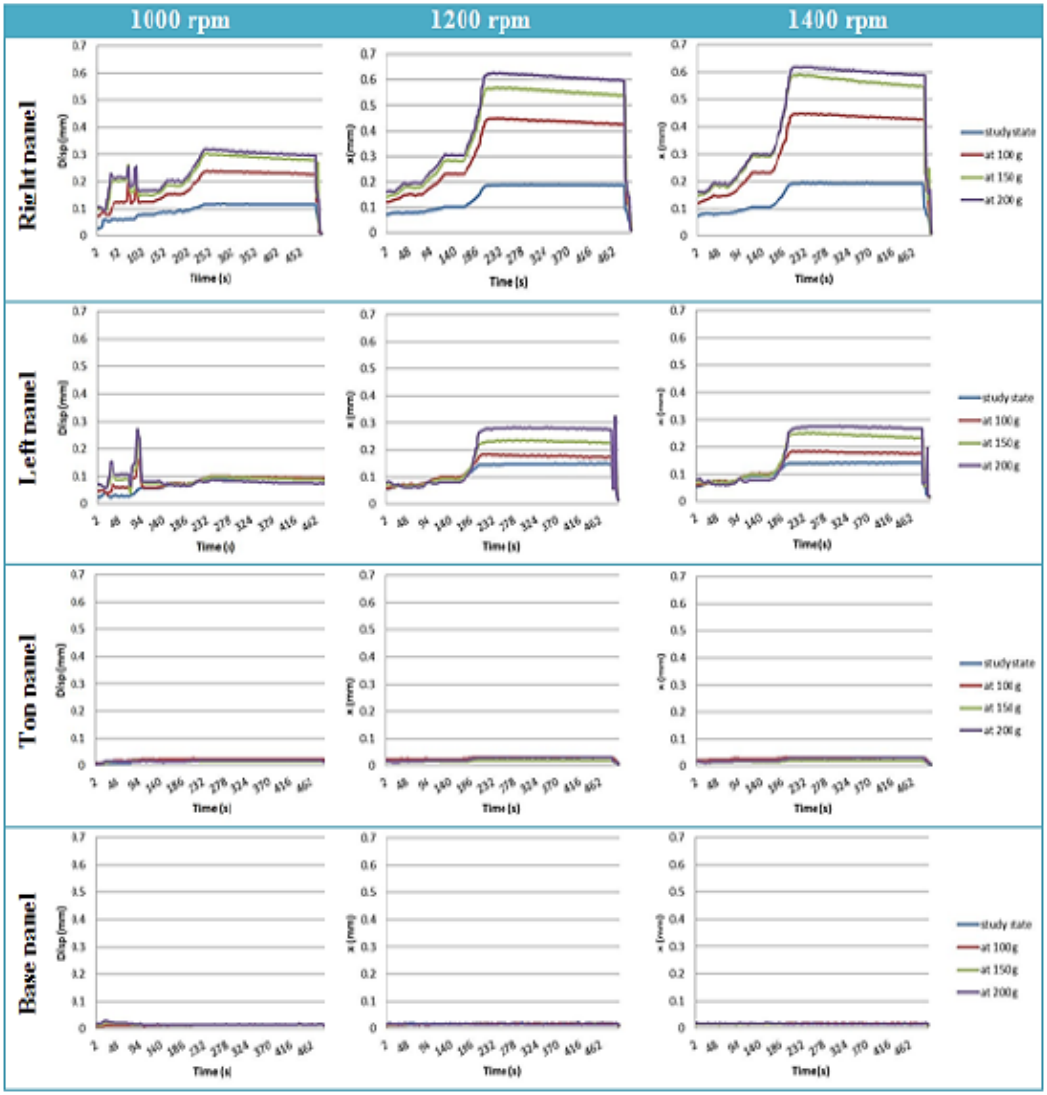
Therefore

$$x = \frac{\left(\frac{mR}{M}\right) r^2}{\sqrt{(1-r^2)^2 + (2\xi r)^2}} \qquad (14)$$

Where,

$$x = steady\ stat\ solution$$

$$\frac{\omega}{\omega_n} = r$$



Amplitude of Vibration in Right, Left, Top and Base panel versus time at different unbalance forces (100,150 and 200) g and different speed (1000, 1200 and1400) rpm

We observe

The vibration amplitude and velocity in the right hand that contains two dampers (right panel) is higher than the other side that contains one damper (left panel).

Increasing vibration amplitude and velocity vibration by increase the proportion of unbalanced mass for each of the right panel and the left panel during the dry speed (1000, 1200 and 1400) rpm

Percentage of vibration amplitude during the spinning cycle (1000,1200 and 1400) rpm with unbalanced masses

Dry Speed (rpm)	Unbalance Mass (g)	Decreasing of Vibration Amplitude (%)		Vibration Amplitude between Right and Left panel (%)
		Right panel	Left panel	
1000	100	49	42	30
	150	62	62	19
	200	63	66	15
1200	100	58	19	59
	150	67	37	58
	200	70	47	55
1400	100	57	23	59
	150	67	43	57
	200	69	48	55

Vibration Amplitude of real parameters and selected parameters

Spinning Speed (rpm)	Unbalance masses (g)	Real Vibration Amplitude (mm)	Selected Vibration Amplitude (mm)	Ratio Improvement (%)
1000	100	0.7998	0.4655	42
	150	1.2101	0.7043	42
	200	1.5927	0.9271	42
	100	0.7915	0.4651	41

Conclusions:

1-It was possible by the proper selection of isolators stiffness, damping coefficient and the drum mass to come down with the vibration amplitude ratio Improvement to 41% at (1200 and 1400) rpm and 42% at 1000 rpm

2- The vibration amplitude and velocity in the side that contains two dampers is higher than the other side that contains one damper

3- Increasing vibration amplitude and velocity vibration by increase the proportion of unbalanced mass during the dry speed (1000, 1200 and 1400) rpm.